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Research paper

Fatigue Life Prediction for Automobile Coil Spring Using Modal Analysis

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Abstract

The aim of this paper is to predict the fatigue life models for a coil spring in vehicle suspension system, which it is a vital part of ground vehicles due to the deflection of the spring is big and continuous. Natural frequencies and mode shapes as known the modal parameters that fatigue life assessment considered as an obstacle in vehicle suspension systems. The finite element analysis performed to obtain the original, mode 1, 2 and mode 3 strain time histories based on the first three modal analyses at each critical area. The objective of this research, to predict the fatigue life in coil spring when it is subjected to free vibration via the hit potholes or bump from any road surfaces. The material design is important to assess the fatigue life, so it has been chosen from the chromium steel, SAE5160 in order to apply frequently in the construction of coil springs. The inverse Fourier transform technique has been utilised in order to produce the strain time history from the power spectral density function. The results show the original and mode 3 signals with amounts of 4.352 and 67.6 cycles have a good agreement with the Morrow model, whereas mode 1 and mode 2 signals indicate 3.7x10-6 and 2.202x105 values in good agreement with the Coffin-Manson model respectively.

***Keywords**: Modal analysis; Coil spring; Fatigue life predictio*

1. Introduction

Modal analysis has been studied extensively in the last few decades. In order to find inherent dynamic specifications of a system such as natural frequencies, damping factors and mode shapes, which determines by modal analysis. Dynamic specifications determine dynamic behaviours to formulate a mathematical model. Modal analysis is a major technology for assessing, improving and optimizing the inherent dynamic characteristics of engineering structures. One of the main obstacles automobile manufacturers to reduce of production body weight on the microscopic scale. For this purpose, the design of complex mechanical, aerospace industry and civil structures require to become increasingly lighter, powerful and more flexible. For example, a coil spring is specified as an elastic part in automotive, in order to resist compression loads and to recover its original shape when the load is removed. Coil springs can also absorb energy from an applied load, which reduces the susceptibility of structures to damage and resonance. Although they can be alternatively under twisting, compression and tension by some loads [1].

Diaz-Cereceda et al. [2] showed vibration transmission in a deterministic approach using modal analysis, in order to modify of elastic joints amongst the floor in the transmission of impact noise. Dziedziech et al. [3] proposed algorithms wavelet-based frequency response function to assess natural frequencies, damping and mode shapes (modal parameters). This method carried out in random impact excitation and signal post-processing based on the crazy climber's algorithm. The results indicated correct modal parameter identification of different noise production mechanisms and noise propagation phenomena on the tyre dynamic behaviour. The results compared with operational modal analysis (OMA) and finite element analysis (FEA), which showed a good agreement with stationary unloaded, stationary loaded and steady-state transport rolling tyre. Prediction of dynamic stresses is an important aspect of engineering structures under random loading, in order to predict fatigue life and safe design

The dynamic stresses are important for fatigue life prediction for better modal responses according to the modal stress superposition and the equivalent behaviour by transferring Power Spectral Density (PSD) into harmonic functions. Moreover, Xie and Xue [4] proposed the new method that the dynamic stresses predicted well the experiment results, which it was successful in high computational performance. Lee et al. [5] presented a computational model for an automotive system to analyse the frequency response parameters by using the modal synthesis technique, in order to obtain vibration modes and forced response characteristics. The results showed a good agreement between the numerical results and a modal test of an experimental up to 300 Hz using the modelling method. In order to analysis techniques of the fatigue failure mode is required to measure the fatigue life. Therefore, fatigue life prediction is developed for test characterisation of components

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in the primary design of product Therefore, fatigue life prediction is developed for test characterisation of components in primary design of product, in order to specific vibration fatigue strength as a vital part in mechanical design [6] Mrsnik et al. [7] improved many frequency domain methods according to numerically simulated signals in vibration fatigue prediction. By comparing different frequency domain methods with results of the experiment, which obtained a good estimation for the Tovo–Benasciutti method in structural dynamics and the automotive industry. Kuznetsov et al. [8] presented a mathematical model to compute the steady-state part of the transmitted vibration. In this model, the vibration analysis is evaluated using the vibration transmitted based on road profile variations to a driver of an automobile. Numerical results demonstrated the effect of different parameters on the level of comfort for the driver because of desirable reduction in body vibrations. Karthik et al. [1] investigated fatigue life analysis by using FEA method on the spring model under variable amplitude loading, in order to display stress and damage values. The results illustrated material SAE 1045-595-QT provided a higher fatigue life than material SAE5160-825-QT SAE1045-450-QT due to the loading sequences is dominantly tensile in nature.

The aim of this study to determine the natural frequencies and mode shapes of the coil spring using modal analysis in each critical strain points. The objective of this study characterises the mode 1, 2 and mode 3 strain time histories to compare with the original strain time history. The modal parameters can be affected on the fatigue life prediction in ground vehicle suspension systems. These modal parameters based on the frequency domain can be converted to the strain time history by using the Inverse Fourier Transform (IFT) technique as time domain analysis. The original strain, mode 1, 2 and mode3 time histories were used as the input to specific software, in order to calculate the accumulation damage and fatigue life amount for coil spring. The strain-time histories predicted the fatigue life based on the Coffin-Manson, Morrow and Smith-Watson-Topper (SWT) strain fatigue life models. The results can be used as the durability test of the structures under variable amplitude loading.

2 Methodology

2.1 Theoretical background

2.1.1 Analytical method of modal analysis

Modal analysis can be defined as a linear combination of the simple harmonic motion of a system, which shows a linear time steady dynamic system by using vibration response. It is called natural modes of vibration. The natural modes of vibration are characterised based upon their spatial distributions and its physical properties such as mass, stiffness and damping. The natural frequency, modal damping and template of characteristic displacement are determined by using terms of modal parameters. Modal analysis is divided into theoretical and experimental techniques. The theoretical modal analysis relies on a physical model of a dynamic system containing physical properties. These properties describe in forms of motion equation of the system. Figure 1, shows the simplify suspension system, which characterised by mass *M*, stiffness *K*, damping *C* and excitation force *F*(t). The motion equation of the system is:

$$M\dot{y}(t) + C\dot{y}(t) + Ky(t) = F(t) \quad (1)$$

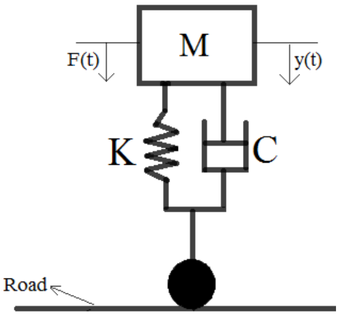


Fig. 1: Scheme of the system with one degree of freedom

By omitting damping, Equation (1) can be written as follows:

$$M\dot{y} + Ky = F(t) \quad (2)$$

By solving differential equations motion above and applying $F(t) = 0$, the equation (3) is:

$$\ddot{y}(t) + \omega^2 y(t) = 0 \quad (3)$$

where:

$$\omega^2 = \frac{K}{M} \quad (4)$$

The solution of equation (3) is in the form:

$$y(t) = Csin(\omega t + \varphi) \quad (5)$$

where, *C*-amplitude (*m*), ω-angular frequency (*rad/s*) and φ-phase angle (*rad*).

By considering the equation (6) and (7) as an oscillation amplitude and phase angle respectively as follows:

$$C = \sqrt{(\frac{\dot{y}}{\omega})^2 + y^2} \quad (6)$$

$$\varphi = arctg\frac{y\omega}{\dot{y}} \quad (7)$$

The angular frequency is named the natural frequency of the system, which corresponds to one of the modes systems. Therefore, equation (3) can be changed to matrix form:

$$[M]\ddot{y}(t) + [K]y(t) = 0 \quad (8)$$

where, *M*-mass matrix (*kg*), *K*-stiffness matrix ($^{N}/_{m}$), y(t)-displacement vector (m) and $\ddot{y}$(t) -vector of acceleration ($^{m}/_{s^2}$).

Displacement and acceleration is given by equation (6) and (7),

$$y(t) = \phi\, e^{i\omega t} \quad (9)$$

$$\ddot{y}(t) = -\omega^2\, \phi\, e^{i\omega t} \quad (10)$$

where ϕ is ean igenvector of the system.

By substituting $\omega^2 = \lambda$ and improving the equation above

$$(K - \lambda M)\phi = 0 \quad (11)$$

Finally, in this study λ is called eigenvalues of the system. It determines a nontrivial solution of the system.

$$det|K - \lambda M| = 0 \quad (12)$$

2.1.2 Fatigue Life Analysis Models

Fatigue life assessment can be determined by one of the Coffin-Manson, Morrow and SWT models because the fatigue life basically derived from this three models, so another model improved of them. The total life is divided to the crack initiation and crack propagation approach (so-called linear elastic fracture mechanics (LEFM)). It calculates the number of cycles, which needs to initiate the small crack. The crack initiation represents elastic-plastic of local stresses or strains. Whereas, the crack propagation approach predicts the pre-existing crack growth rate and estimates the number of loading cycles to grow the critical size of crack when the failure suddenly will accrue [1]. In the last few decades, research studies have been accomplished for mean stress effect on fatigue life according to strain life model, when the elastic-plastic stress-strain range is based on the cyclic response of the material. The mean stress and mean strain effect are important to fatigue life prediction because the engineering components are mostly subjected to cyclic loads. The correlation between the total strain amplitude $(\frac{\Delta\varepsilon}{2})$ and fatigue life (N_f) can be represented by the Coffin-Manson model [9] in equation (13)

$$\frac{\Delta\varepsilon}{2} = \varepsilon_a = \frac{\Delta\varepsilon^e}{2} + \frac{\Delta\varepsilon^p}{2} = \frac{\sigma'_f}{E}\,(2N_f)^b + \varepsilon'_f\,(2N_f)^c \tag{13}$$

where, σ'_f is the fatigue strength coefficient, E is the modules of elasticity, ε'_f is the fatigue ductility coefficient and c is fatigue ductility exponent. The equation (13) can be modified by the mean stress effect parameter, in order to assess the fatigue life at zero mean stress. By modifying the elastic part of the strain life model using the mean stress effect (σ_m), therefore the Morrow proposed the mean stress effect as follows [1]

$$\varepsilon_a = \frac{\sigma'_f - \sigma_m}{E}\,(2N_f)^b + \varepsilon'_f\,(2N_f)^c \tag{14}$$

The SWT strain life model suggested another mean strain effect (the SWT mean stress correction model), therefore it is mathematically expressed in equation (15), where σ_{max} is the maximum stress [10]:

$$\sigma_{max}\varepsilon_a = \sigma_{max}\frac{\Delta\varepsilon}{2} = \frac{(\sigma'_f)^2}{E}\,(2N_f)^{2b} + \varepsilon'_f\sigma'_f\,(2N_f)^{b+c} \tag{15}$$

The Palmgren-Miner method as known as the linear damage accumulation rule. It is expressed to assess the fatigue life, which creates the total damage. This method is very effective in the automobile industry in order to calculate fatigue damage of structure. The linear damage accumulation rule can be determined using the sum of total partial damage as follows [11]:

$$D = \sum_{i=1}^{n} \frac{n_i}{N_{f_i}} = 1 \tag{16}$$

where D is the accumulated damage, n_i is the number of applied cycles and N_f is the number of constant amplitude cycles to failure. The structure is failed when the total damage reached to the value of 1.

2.2 Materials and Methods

Since automotive suspension components more tend to mechanical failure due to fatigue damage resulting from the variable amplitude loading under service condition. Figure 2, illustrates the flowchart steps to get the objective of the current study. A coil spring was selected, in order to investigate the correlation between fatigue life and modal analysis, which Table 1, shows the first three modes and natural frequencies. In this study, the strain time history considered as the input data that this repeated loading can be known as the fatigue signal. The first step in this process was to simulate the coil spring based on modal analysis by using FEA, in order to characterise natural frequencies and mode shapes at each critical area. Figure 3, shows the first three modal analysis of the current study. By converting to the strain time history using IFT technique from the PSD at the second stage. The third step illustrates the extraction mode 1, 2 and mode 3 and the original strain time histories from the previous step based on the time domain as the input fatigue life analysing. The Coffin-Manson, Morrow and SWT models exist to compare the results of fatigue life and damage between the original strain signal and mode 1, 2 and mode 3 strain signals in the fourth step. The fatigue life prediction can be determined if the results are satisfactory and accurate from previous steps. Finally, the process predicts the fatigue life model at the fifth step. The material properties are chosen from the chromium steel, SAE5160, in order to use frequently in the construction of coil springs. Table 2, showed the material properties and their definitions, which is selected for the automotive suspension component.

Table 1 : The first three modes and natural frequencies

Number of modes	Frequencies (Hz)
Mode 1	8.5
Mode 2	8.55
Mode 3	43.7

Table 2: Mechanical properties of the SAE5160 steel [1]

Properties	SAE5160 steel
Yield strength (MPa)	1070
Ultimate tensile strength (MPa)	1550
Material modulus of elasticity (GPa)	207
Fatigue strength coefficient (MPa)	2063
Fatigue strength exponent	-0.08
Fatigue ductility exponent	-1.05
Fatigue ductility coefficient	9.56
Cyclic-strain hardening exponent	0.10
Cyclic strength coefficient	2000

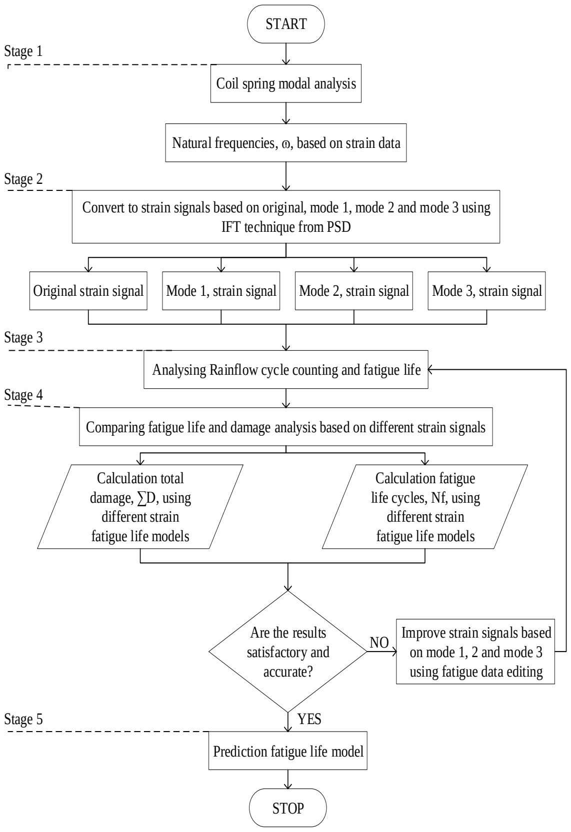


Fig. 2: The flow chart of the study

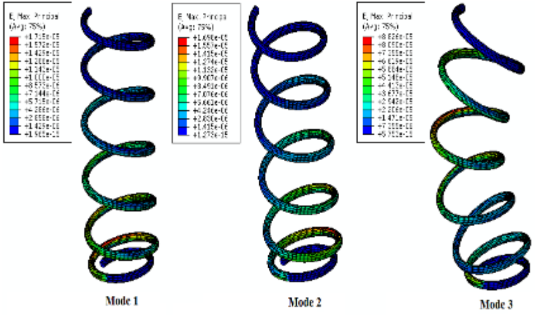


Fig. 3: The first three modal analysis of coil spring analysis

1. Results and Discussion

Coil spring is subjected to random excitation by the wheels on the road. The signals of random acceleration fall out in a term. This signals are usually Gaussian and linear, which can be fundamental criteria of fatigue vibration. These signals assist to obtain the fast Fourier transform (FFT) and PSD functions [12]. Figure 4, shows both damage and strain time history in a plot together. By considering this two parameters the high strain value aid to fatigue damage due to existing large values of strain in the overall signal. As seen in Figure 4, the lower energy contribute to lower fatigue damage, whereas, the high amplitudes detect the higher energy from the potential fatigue damage signals. It indicated the higher energy or higher amplitude causes higher fatigue damage. Therefore, as discussed earlier above this plot able to show which signal contribute to fatigue damage and can be helpful again in diagnosing fatigue problems, for example, what causes the damage, what is happening when damage fatigue cycles occur and how the part being is used at that time.

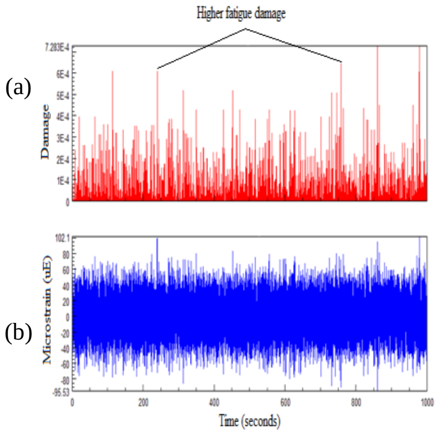


Fig. 4: Plots of fatigue analysis. (a) damage magnitude (b) strain time history

In order to predict the fatigue life estimation based on the time domain, Rainflow cycle counting technique and fatigue damage rules must be used. For this purpose, the time domain analysis by using Rainflow cycle plots and damage accumulation model can be applied. Mechanical parts of automotive are frequently subjected to apply random loads. Since the signal time history cannot be applied instantly fatigue calculation due to its complex. Therefore, by decomposing the signal time history the Rainflow cycle counting technique can be utilised. So in this case, there are some strategies of cycle counting, which lead to a different count.

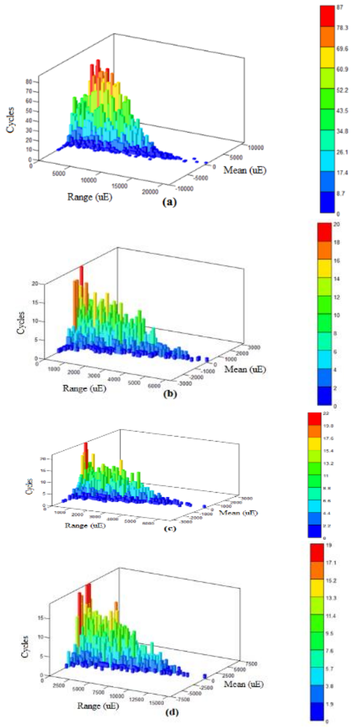
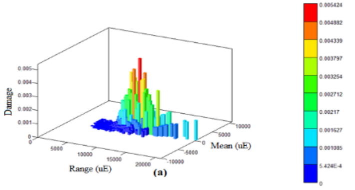


Fig. 5: Cycle histogram distribution for (a) original strain signal (b) mode 1 strain signal (c) mode 2 strain signal (d) mode 3 strain signal



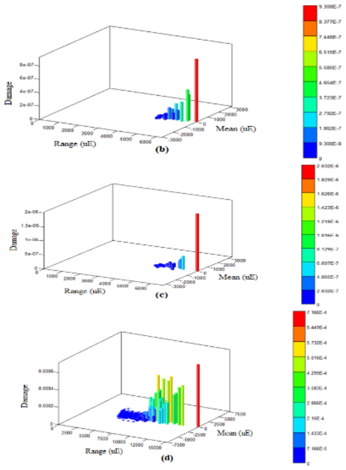


Fig. 6 : Damage histogram distribution for (a) original strain signal (b) mode 1 strain signal (c) mode 2 strain signal (d) mode 3 strain signal

Figure 5, demonstrates Rainflow cycle counting results based on the Coffin-Manson, Morrow and SWT strain life models at the critical site of coil spring respectively. In this plot, x, y and z-axes illustrate load range, load mean and a number of cycles for each cycle in the time history. The majority of the damage that is characterised by Rainflow cycle, which presented an idea on it. This idea expressed many cycles appeared by the strain with low cycles are located on the left of the plot, against few cycles from high strain range on the right. The height of each bar Indicative number of cycles that at specific of strain range and mean. Hence, the results represented that most of the damage occurred at the high strain ranges [13]-[14]. The simulation process provided fatigue damage distribution for each cycle as indicated by the corresponding three-dimensional damage histograms in Figure 6. The plot shows a pattern of the majority of the damage based on original, mode 1,2 and mode 3 strain time histories. A few cycles contributed to high strain range, meanwhile many cycles were from low strain range. The height of each bar illustrates the number of cycles at strain range and mean. The results of the illustrated damage distribution are considered in the high range. Therefore, each high damage characterises by increasing the range.

Table 3 : The fatigue life predictions

Strain signals		Coffin-Manson	Morrow	SWT
Original	Fatigue life (cycles)	4.343	4.352	4.348
Mode 1	Fatigue life (cycles)	2.67×10^{5}	2.63×10^{5}	2.57×10^{5}
Mode 2	Fatigue life (cycles)	2.202×10^{5}	2.001×10^{5}	1.86×10^{5}
Mode 3	Fatigue life (cycles)	67.1	67.6	67.5

Table 4 : The fatigue damage predictions

Strain signals		Coffin-Manson	Morrow	SWT
Original	Fatigue damage	2.302×10^{-1}	2.298×10^{-1}	2.3×10^{-1}
Mode 1	Fatigue damage	3.7×10^{-6}	3.8×10^{-6}	3.9×10^{-6}
Mode 2	Fatigue damage	4.5×10^{-6}	5×10^{-6}	5.4×10^{-6}
Mode 3	Fatigue damage	1.49×10^{-2}	1.48×10^{-2}	1.48×10^{-2}

The results of the analysis are presented when the natural frequencies are increasing the fatigue life decreasing and damage growing also. Table 3 and 4, show the results for the fatigue life and damage, where the results obtained based on the Coffin-Manson, Morrow and SWT strain fatigue life models for original, mode 1, 2 and mode 3 strain time histories. The mode 1, 2 and mode 3 depend on the first bending, first twisting and second bending respectively. According to Table 1 and 2, strain time history at 43.7 Hz generates the highest damage about 1.5×10^{-2}. The mode 2 and mode 1 determined 5.4×10^{-6} and 3.9×10^{-6} another highest damage values, where the frequencies are 8.55Hz and 8.5 Hz respectively. By comparing the results of fatigue and damage in Table 3 and 4, between mode 1,2 and mode 3 the lower frequency indicates the lower damage and the higher life prediction. The Coffin-Manson model has a good result in the mode 1 and mode 2 due to predicting only the fatigue life at zero mean stress. Whereas, The original strain time history and mode 3 show good results in the Morrow model where the elastic strain values dominate or the mean stress has a significant effect on longer lives. It is clear in Table 3 and 4 that the original strain time history has the lowest and highest fatigue life and damage respectively due to the original strain signal is included from mode 1, 2 and mode 3 strain signals together.

Table 5: Statistical analysis for the comparison of the strain-life models prediction

Strain signals		Coffin-Manson	Morrow	SWT
Original	Mean value	4.605×10^{-6}	4.595×10^{-6}	4.6×10^{-6}
	Standard deviation	8.61×10^{-6}	8.58×10^{-6}	8.58×10^{-6}
Mode 1	Mean value	7.5×10^{-6}	7.6×10^{-6}	7.8×10^{-6}
	Standard deviation	1.2×10^{-6}	1.3×10^{-6}	1.4×10^{-6}
Mode 2	Mean value	9.1×10^{-6}	10×10^{-6}	10.7×10^{-6}
	Standard deviation	2.2×10^{-6}	2.4×10^{-6}	2.5×10^{-6}
Mode 3	Mean value	2.98×10^{-6}	2.95×10^{-6}	2.96×10^{-6}
	Standard deviation	1.63×10^{-6}	1.62×10^{-6}	1.62×10^{-6}

Table 5, shows mean value and standard deviations, this table indicates a good rule for statistical quantities as a briefing of central trend and extension to predict for the models. In order to higher accuracy of the proposed model the lower mean and standard deviations are needed to predict [9]-[10]-[15]. As clearly seen in table 5, in mode1 and mode 2 strain signals the SWT model tends to overestimate fatigue lives with the largest mean and standard deviation values and the Morrow model provides conservative predictions with positive mean values and lower standard deviations as well. Whereas, in the original strain signal and mode 3 strain signal the Coffin-Manson model and the SWT model were overestimated and conservatives. Ince et al. [9] and Zhu et al. [10] investigated statistical analysis for comparison of prediction models, where the results have shown good agreement with experiment.

Hence, the preference chooses one model toward to the other models is difficult to categorically. Consequently, when the mean stress of loading is zero the Coffin-Manson model can be dominated [16]. The Morrow model can be utilised for compressive mean stress, particularly vehicle components that subjected to compressive loading frequently [10]-[13]. Finally, the SWT model has introduced a flow that more conservative in life estimation for the tensile mean stress [13].

2. Conclusion

This paper investigated the comparative study of total fatigue life and damage. A coil spring of vehicle suspension, for example, is defined, which considered the Coffin-Manson, Morrow and SWT strain fatigue life models to assess the fatigue life and damage of typical vehicle components. The first three natural frequencies obtained 8.5 Hz, 8.55 Hz and 43.7 Hz for mode 1, 2 and mode 3 respectively. The original time history compared with mode 1, 2 and mode 3 strain time histories. All strain signals that were explained in this research captured from the frequency domain, in order to analyse based on time domain using IFT technique in each strain time history. Based on the time domain signal fatigue damage can be characterised by Rainflow cycle counting and strain life relationship. According to the time domain signal, the high amplitudes can be easily identified as the potential fatigue damage signals.

As summarise, the results have shown The Coffin-Manson model has a good agreement for mode 1 and mode 2, strain signals by the amount of 2.67×10^{5} and 2.202×10^{5} cycles in fatigue life and 3.7×10^{-6} 4.5×10^{-6} in fatigue damage. Whereas the Morrow model indicated a good agreement for both the original strain and mode 3 strain signals by amount of 4.348 and 67.5 cycles in fatigue life and 2.298×10^{-1}, 1.48×10^{-2} in fatigue damage values. Therefore, the SWT model has shown tend to overestimate fatigue lives with the largest mean about 7.8×10^{-6}, 10.7×10^{-6} and standard deviation around 1.4×10^{-6}, 2.5×10^{-6} magnitudes based on mode 1 and mode 2 respectively. Meanwhile, the Morrow model has shown conservative predictions in lower standard deviation about 1.3×10^{-6}, 2.4×10^{-6} and mean 7.6×10^{-6}, 10×10^{-6} values respectively. Original and mode 3, strain signals have overestimated and conservatives for the Coffin-Manson and SWT models.

Acknowledgements

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